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An investigation of lubrication oil film breakdown pressures under steady and intermittent loading

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AN INVESTIGATION OF LUBRICATING OIL FILM BREAKDOWN PRESSURES UNDER STEADY AND INTERMITTENT LOADING

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AN INVESTIGATION OF LUBRICATING OIL FILM BREAKDOWN PRESSURES UNDER STEADY AND INTERMITTENT LOADING

A Thesis

By

Lieutenant (junior grade) D. H. CONE, U. S. NAVY
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and

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Submitted in partial fulfillment
for the degree of
MASTER OF SCIENCE

June, 1932

Approved: May 19, 1932 . . . .

Louis J. Bradford . . . . .
Professor of Machine Design

Thesis
C7

Introduction

The investigation of breakdown pressures of lubricating oil films was an attempt to establish a definite relation between the effects of steady and intermittent loads on a bearing. The work was extended to include the correlation of as many of the variables which enter into the establishing and maintaining of an oil film as was possible. Such an investigation had been instituted in 1931 by Lieutenant B. L. Blue, U. S. Navy, and Lieutenant (junior grade) R. L. Ewart, U. S. Navy. Credit is due them for their pioneering work in the construction of the basic machine and initiating the test. Various recommendations by Lieutenants Blue and Ewart for modifications in the machine and procedure have been carried out insofar as time and funds have permitted.

It was necessary to obtain additional equipment and replace some of the original which was not available for our use during this current year. Considerable time was spent in locating and obtaining this equipment, in manufacturing various parts of the apparatus, and in making the various modifications.

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Description of Apparatus

The machine proper for testing the breakdown pressure of oil films consists primarily of a test bearing and shaft, a loading mechanism by which the load is applied to the test bearing, a means of varying the magnitude of the load, a mechanical method of alternately applying and releasing the load, and a means of furnishing a continuous and constant supply of lubricating oil to the test bearing.

TEST BEARING.

The test bearing is a bronze shell with a babbitt lining approximately 0.125" in thickness. The shell was originally 4 1/4" in diameter and 2" in length, but has been somewhat flattened on the top to provide a convenient surface for application of the load. The overall diameter has been further increased by the construction of a water jacket of sheet copper material, having a water space of 1/2". The jacket extends around the two sides of the circumference as shown in Figure 6. It was not practical to extend the jacket completely around the circumference due to the necessity of applying the load to the bearing on the top, and of introducing the lubricating oil into the bearing at the bottom. A thermometer well was drilled radially into the shell, and tangent to and in

contact with the babbitt as far as practicable. The babbitt was turned to a nominal diameter of 1.998" and reamed to give a .006" diametral clearance on the test shaft. An oil groove 1 3/4" long, 3/16" wide, and 1/8" deep was cut axially along the bottom of the bearing at the point where the lubricating oil is introduced.

TEST SHAFT.

The test shaft is of case-hardened steel, ground and polished to a diameter of 2.0005" along its length. It is 21" long, both ends extending beyond its support bearings sufficiently to permit the installation of a pulley on one side and the use of a hand tachometer on the other. The shaft is not restrained axially, but is free to find its own running position in the bearing. It is belt-driven by a 5 H.P., 120 volt, shunt wound, direct current, motor, speed control of which is obtained by means of two portable lamp banks in the field circuit.

LOADING MECHANISM.

The loading mechanism consists of several parts, namely, a load bar, four 3/4" steel tie-rods, and a strong back which receives the load transmitted by the tie-rods from the load bar. The strong back loads the bearing which is located at its central point. An attempt was made to avoid eccentricity of loading by using a 3/8" steel ball between the strong back and test bearing. It was hoped

that the "point loading" would not cause canting of the bearing on the test shaft with a possible resultant local breakdown of the oil film. In addition to the ball, three 1/8" strips of bakelite were inserted between the strong back and bearing for electric insulation. Two stud bolts from the bearing extend through the strong back acting as its vertical guides, and also serve to prevent the test bearing from rotating with the shaft.

The strong back is made of mild steel 4" x 2" x 13" and is sufficiently heavy so that its deflection due to center loading is negligible. It is drilled at each end to receive the tie-rods and transmitted load from the load bar. It is also drilled to permit the passage of two oil cup feeds to the test shaft support bearings and a rubber hose jumper-connection between the water jackets on the two sides of the test bearing.

The load bar is of mild steel material 2" x 1" x 14". It was annealed at 1475°F. in the electric furnace for ten minutes and then drawn, after an oil quench, at 650°F. for thirty minutes to obtain sufficient strength to carry a center load of 5000 pounds without taking a permanent set. It was next calibrated in a hand-operated, tension-testing, machine to obtain a load-deflection curve. A dial indicator was suitably mounted on a flat plate directly under the point of loading so that deflections

of the bar under the various loadings could be accurately measured. From these deflections and the applied load, as obtained from the testing machine, a load deflection curve was constructed. The curve was modified to include the weight of the loading mechanism, which was 58 1/2 pounds, so that the actual bearing pressure in pounds per square inch of projected area could be obtained for any deflection. This curve is shown in Figure 4. Supports at each end of the load bar transmit the load to the tie-rods.

The point of loading is the head of a 3/8" hexagonal bolt secured on the bottom of the lever bar. The latter is a specially machined steel shape from original mild steel stock 2" x 2 1/2" x 15". It is fitted at one end with a 2" case-hardened roller follower which remains in contact with a rotating cam. The lever is supported by two springs with sufficient tension to insure contact between cam and follower. It is pivoted at the other end on a piece of 2" round steel stock so that a ratio of the distance from the pivot to the roller to the distance from the pivot to the load bar is 5 to 1. In this way the load on the load bar is five times the pressure which the cam exerts on the follower.

The cam is case-hardened, medium carbon steel. It is 2" wide and is designed to give a lift of 1/4" over

an arc of 180 degrees. The cam is rotated through a Chevrolet transmission by a 2 H.P., 120 volt, shunt wound, direct current motor. The latter is fitted with a field rheostat for speed control. As the transmission allows speed ratios of 3:1, 1 1/2:1, and 1:1, a wide range of speed for the cam is available.

ELECTRICAL CIRCUIT.

The closing of an electrical circuit by oil film breakdown determines the point of breakdown conditions. The circuit consists of four dry cells connected in parallel and in series with a small 6 volt test lamp, an ammeter and a knife switch. One lead from this apparatus is taken to one of the test bearing guide bolts above the strong back. The other goes to a suitably mounted copper wire which remains in contact with the rotating test shaft. As long as an oil film is maintained between the shaft and test bearing, the electrical circuit is interrupted and no current can flow. At the instant of oil film breakdown, the circuit is complete, the lamp lights, and the ammeter needle is deflected.

LUBRICATING OIL SUPPLY.

The lubricating oil supply is from a tank located 25 feet above the machine. The oil, which is previously centrifuged, is received at the bearing through a filter at a pressure of 9.5 pounds per square inch. Time per-

mitted the test to be made with only one lubricant, the characteristics of which are here given.

TEXACO REGAL "C" OIL

Gravity	20.1
Flash point (Cleveland Open Cup Method)	365°F.
Fire point	415°F.
Viscosity in Saybolt Seconds	
100°F.	324 seconds
130°F.	137 seconds
70°F.	1090 seconds
Pour point	0°F.

See Figure 3 for the temperature-viscosity curve for this oil.

Procedure

The load bar deflection was obtained from the load-deflection curve for the bearing pressure to be tested. The cam was rotated to give the maximum deflection of the lever bar. The adjusting nuts on the tie-rods were then set to give the desired deflection of the load bar. When adjustments were satisfactory the test shaft was rotated by its motor. Speed was increased until the dying out of the test lamp or zero deflection of the ammeter indicated that an oil film had been established. The shaft was allowed to run to insure constant conditions in the bearing, and then gradually slowed until breakdown of the oil film occurred. This was evident by the deflection of the ammeter or flickering or glowing of the test lamp. A steady glow, barely visible, was arbitrarily assumed to be the point of breakdown of the oil film. Test journal r.p.m. and temperature of the bearing was obtained for this condition. The procedure was repeated several times until journal r.p.m. had been satisfactorily checked.

After obtaining breakdown data for steady load condition, the cam operating motor was started, and the load applied intermittently. With each revolution of the cam the load was alternately applied and relieved. The

journal speed was increased until a film was established, and then gradually slowed as before until film breakdown occurred.

Data was taken for bearing pressures between 100 and 500 pounds per square inch of projected bearing area for steady and intermittent loading. Test runs were made to determine journal speeds at film breakdown points for various frequencies of loading. Also data was obtained in an effort to correlate bearing temperature and oil viscosity with journal speed for a constant loading.

Oil Viscosity vs. Journal Velocity for Steady Loads

It is well known to all who have observed heavy machinery start from rest and run slowly that frictional resistance is great. This is evident from the groans emanating from the bearings, and the jerky irregular motion of the moving parts. Under these conditions there is metal to metal contact and interlocking of the minute irregularities of the surfaces. Boundary friction is occurring. As speed is increased, the groans become fewer and less intense, and finally the movement is smooth, even, and noiseless. The viscous film of oil has been formed, perfect lubrication is obtained, and a viscous friction condition is the result. Relative movement of the two surfaces has built up an oil film due to viscous drag, until they are entirely separated. Under such conditions friction is that due to the resistance to shear offered by the lubricant. The value of this resistance depends upon

1. Viscosity of the lubricant
2. Relative speed of the surfaces
3. Area of the surfaces
4. Thickness of the oil film.

Since the friction of journals is practically independent of the load when the speed is sufficient to

maintain a pressure film between the two surfaces[1], it would be valuable to know the relation of these variables one to another in establishing and maintaining the oil film. An attempt was made to correlate the first two, namely, the viscosity of the lubricant and relative speed of the surfaces. Time did not permit an investigation of the effect of varying surface area and thickness of oil film. This might easily be done by using test bearings of various lengths, and varying radial clearance respectively. The results of the investigation are shown in Figure 1. A logarithmic plot of these data shows that journal speed varies inversely as (Viscosity Saybolt Universal) $^{1.15}$ or Journal r.p.m. = $\left(\frac{1}{\text{Viscosity}}\right)^{1.15}$ for a constant steady load.

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[1] Lubrication and Lubricants. p. 107. Archbutt and Deely.

Results

The results obtained have been divided into two relationships shown graphically by the curves of figures 1, 2, and 2A.

1. Figures 2 and 2A represent the direct results of the basic investigation. In Figure 2A oil film breakdown pressures in pounds per square inch of projected bearing area are plotted against corresponding journal speeds in revolutions per minute. These points were taken with the machine operating at a cam frequency of 100 per minute, and at bearing temperatures varying from 60° to 70°F. This curve indicates that the load carrying capacity of an oil film within a bearing of the type used under conditions of steady loading is greater than that of an oil film under conditions of intermittent loading. It is noted that the increase in breakdown pressures versus journal speeds is a straight line relationship in each of the two types of loading. The curves being practically parallel show that the breakdown pressure under steady loading exceeds that of the intermittent loading by about 45 pounds per square inch. In Figure 2 the breakdown pressures in pounds per square inch of projected bearing area are plotted against the function "$\frac{ZN}{P}$", the symbols of which have the following significance:

Z - absolute viscosity in centipoises
N - journal speed in revolutions per minute
P - bearing pressure in pounds per square
inch of projected area

(The absolute viscosity in centipoises was obtained by use of curve 3A, the temperature of the bearing being assumed to be the same as the temperature of the oil film. The error incurred by this assumption is believed to have no effect on the comparative value of the results.) Figure 2 shows the relationship of the breakdown pressures for the two types of loading with the temperature factor corrected for. That is, it eliminates the possibility of distortion due to changes in viscosity resulting from the 10°F. variation in bearing temperature. This curve also shows the breakdown pressure to be greater for steady loading than for intermittent loading.

2. In Figure 1 viscosity, Saybolt Universal, and bearing temperatures in degrees Fahrenheit were plotted against journal velocity in feet per minute. This curve shows that increases in bearing temperatures indicate lower viscosities and require greater journal velocities to maintain the oil film under a constant load. (In this case, the load was 280 pounds per square inch of projected area)

Discussion of Results

1. The results indicated in Figures 2 and 2A are contrary to generally accepted theories as to how the breakdown pressures under steady and intermittent loading should compare. Archbutt and Deeley make the following statement, "In some cases the loads upon bearings are by no means constant, for the faces often alternately approach and recede from each other. When this is the case, and the alternation is very rapid, the bearing will carry a very great weight, for at each alternation the pressure is completely relieved, and the oil 'trapped' cannot be expelled during the short time the load rests on the bearing."[1]

It is probable that in the case of intermittent loading the load as applied to the bearing became eccentric due to a slight angular movement of the loading mechanism in the wake of the lever bar. This would cause the oil film to break down more quickly than if it were applied through the center of the bearing, as in the case of steady loading which existed, more nearly duplicated the conditions found in many engineering uses of this type of loading, as in crank pin and wrist pin bearings.

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[1] See "Lubrication and Lubricants" by Archbutt and Deeley.

2. A study of Figure 1 shows that variations in the temperature of the oil film and consequent changes in its viscosity have a marked effect upon the carrying power of the film.

Recommendations

It is recommended that further investigations of the behavior of an oil film in a bearing under steady and intermittent loads be conducted. In connection therewith the following suggestions are submitted:

1. The rigidity of that part of the loading mechanism including the load bar, the rods, and strong back should be improved to prevent angular motion of those parts as the load is applied. This should eliminate excessive vibration and the possibility of eccentric loading.

2. Finer speed adjustment of both the journal and cam operating motors should be secured. Also a wider range of speed control for the journal operating motor is necessary in order to investigate conditions of loading greater than 600 lbs. per sq. in.

3. The effect of frequency of loading upon the breakdown pressures is still undetermined and offers an interesting field for further research.

4. The effect of time of application of the load, and the use of various types of oils suggest other lines of investigation.

5. It would be interesting, if possible, to analyze the breaking down process within the film from the point

of first metal to metal contact until maximum breakdown has occurred, for the two conditions of loading, and to determine at what stage of the process scoring of the bearing or seizure is apt to result.

This analysis might be made by use of a milliammeter in the indicator circuit, providing delicate speed and load control have been secured and there is very little vibration in the apparatus. The bearing should be removed and inspected for scoring after each desired point of breakdown has been reached.

Acknowledgment

Acknowledgment is made primarily to the Navy Department, which through the Postgraduate School of the U. S. Naval Academy made available both the time and the funds for this research work, and which also supplied the two necessary motors.

We are greatly indebted to L. J. Bradford, Professor of Machine Design at The Pennsylvania State College for his many helpful suggestions.

We also wish to express gratitude to L. A. Doggett, Professor of Electrical Engineering; F. C. Stewart, Associate Professor in Charge of the Mechanical Engineering Laboratory; and to the Department of Metallurgy for their aid and cooperation.

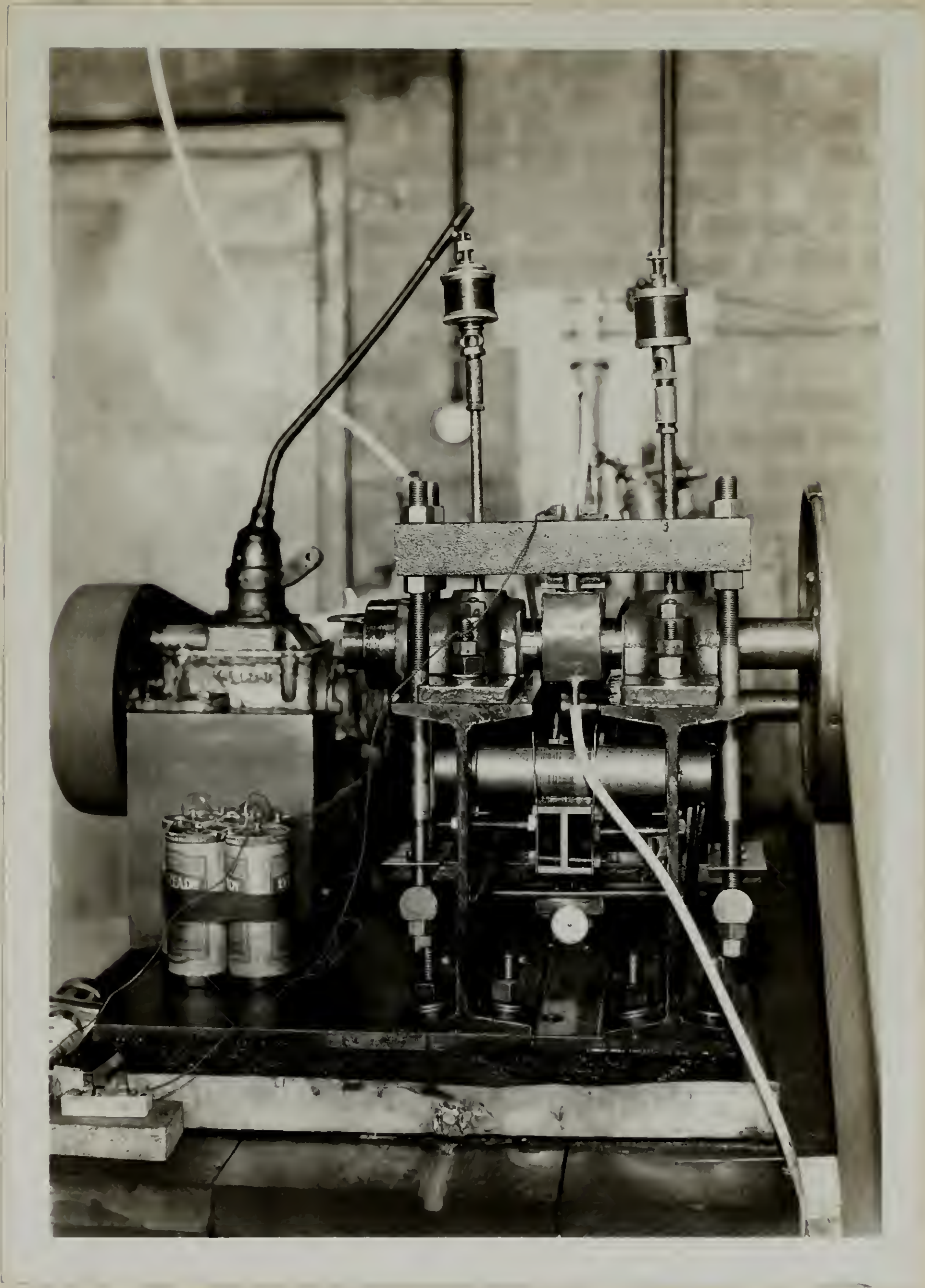


PLATE 1

END VIEW OF MACHINE

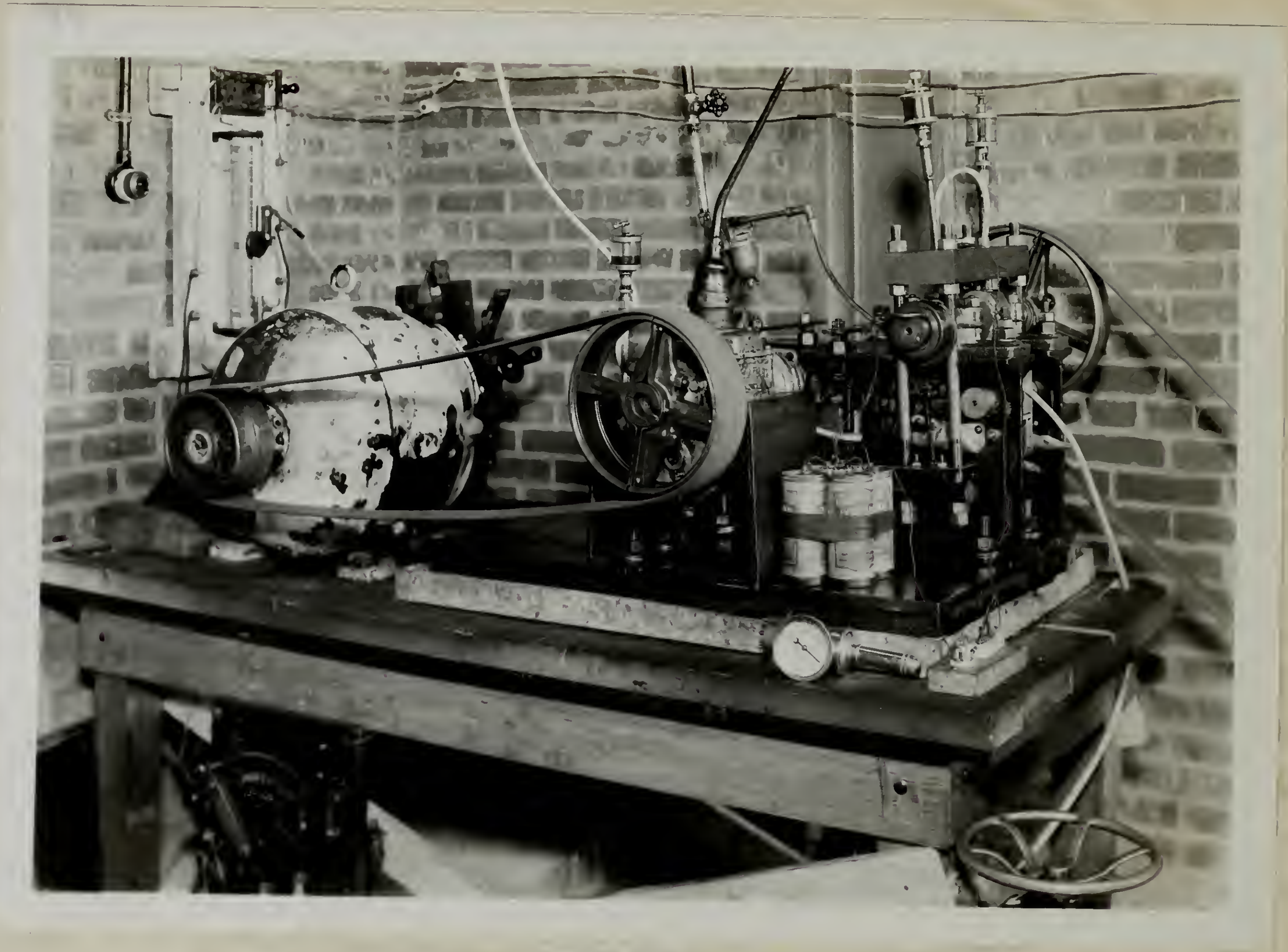


PLATE 2

SIDE VIEW OF MACHINE AND CAM OPERATING GEAR

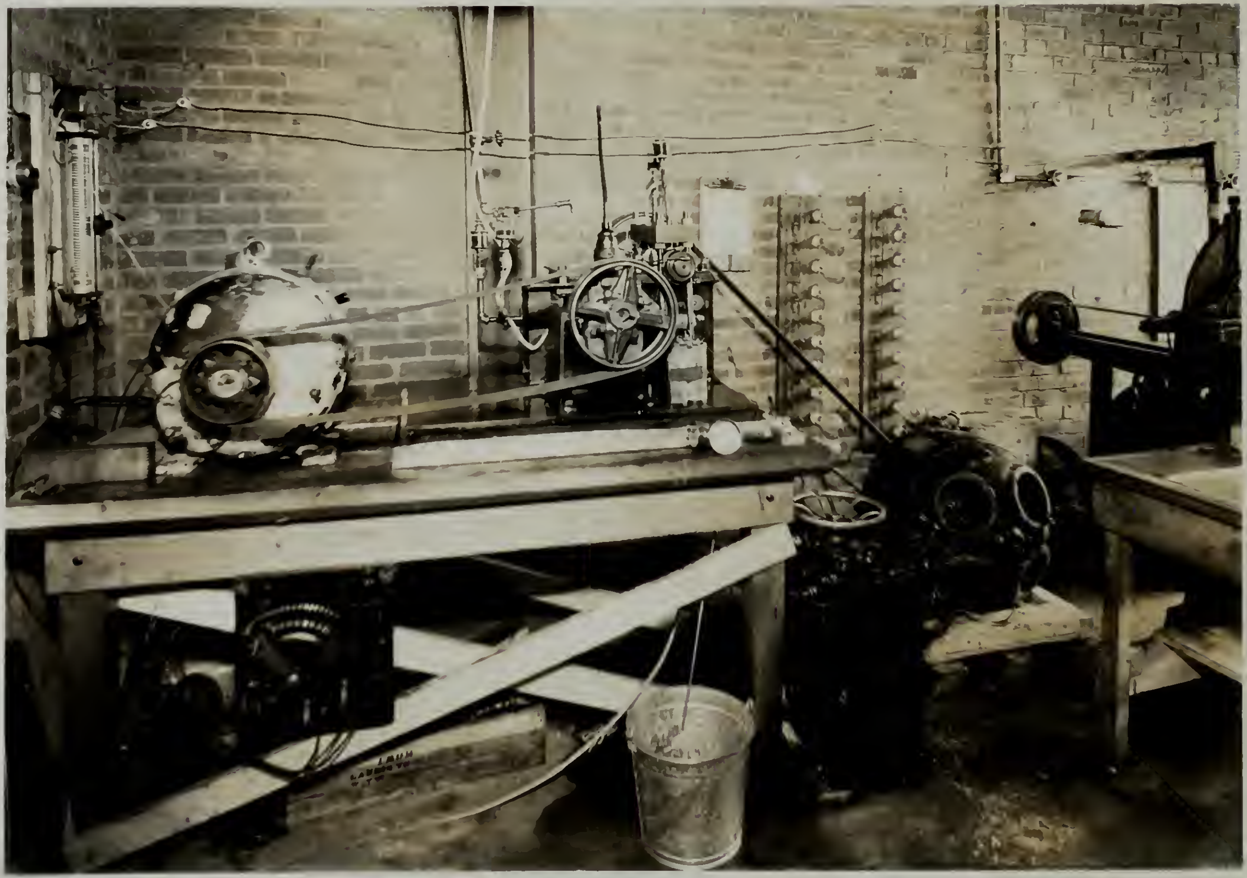
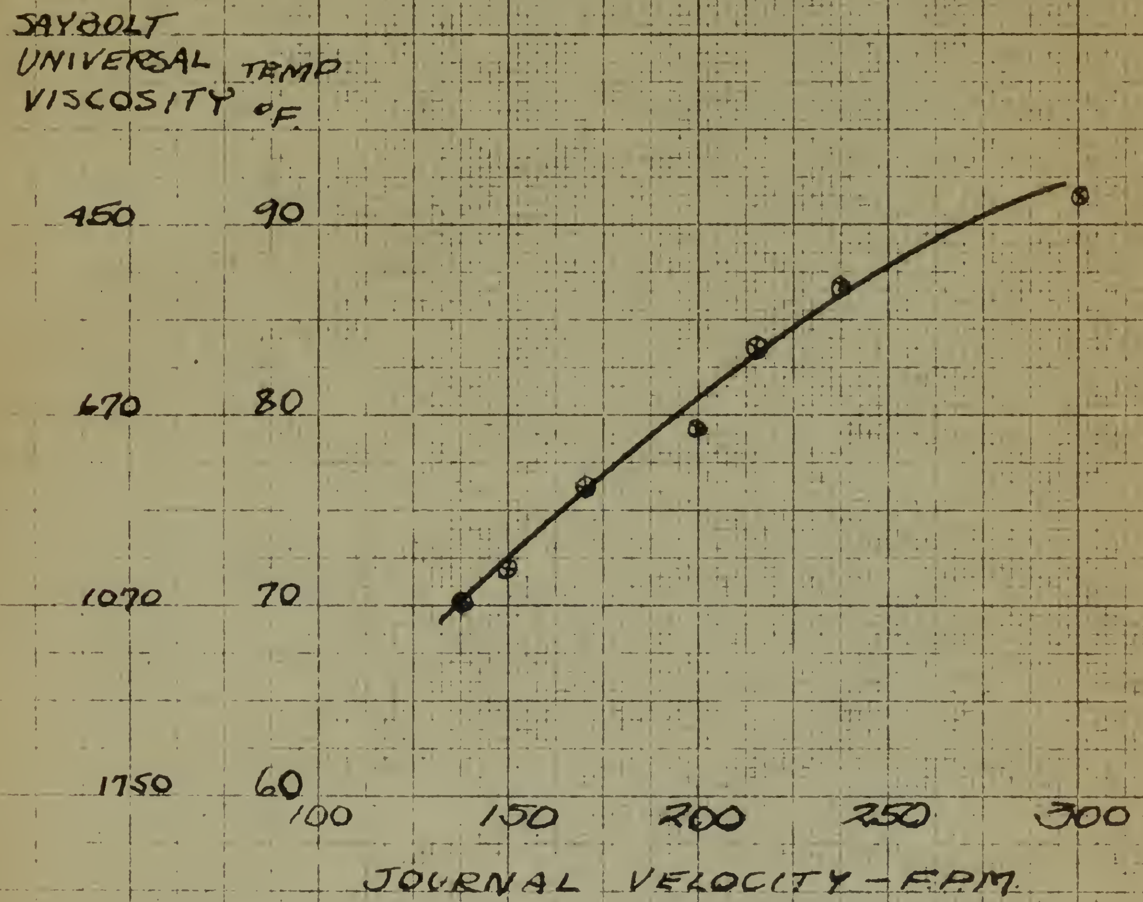


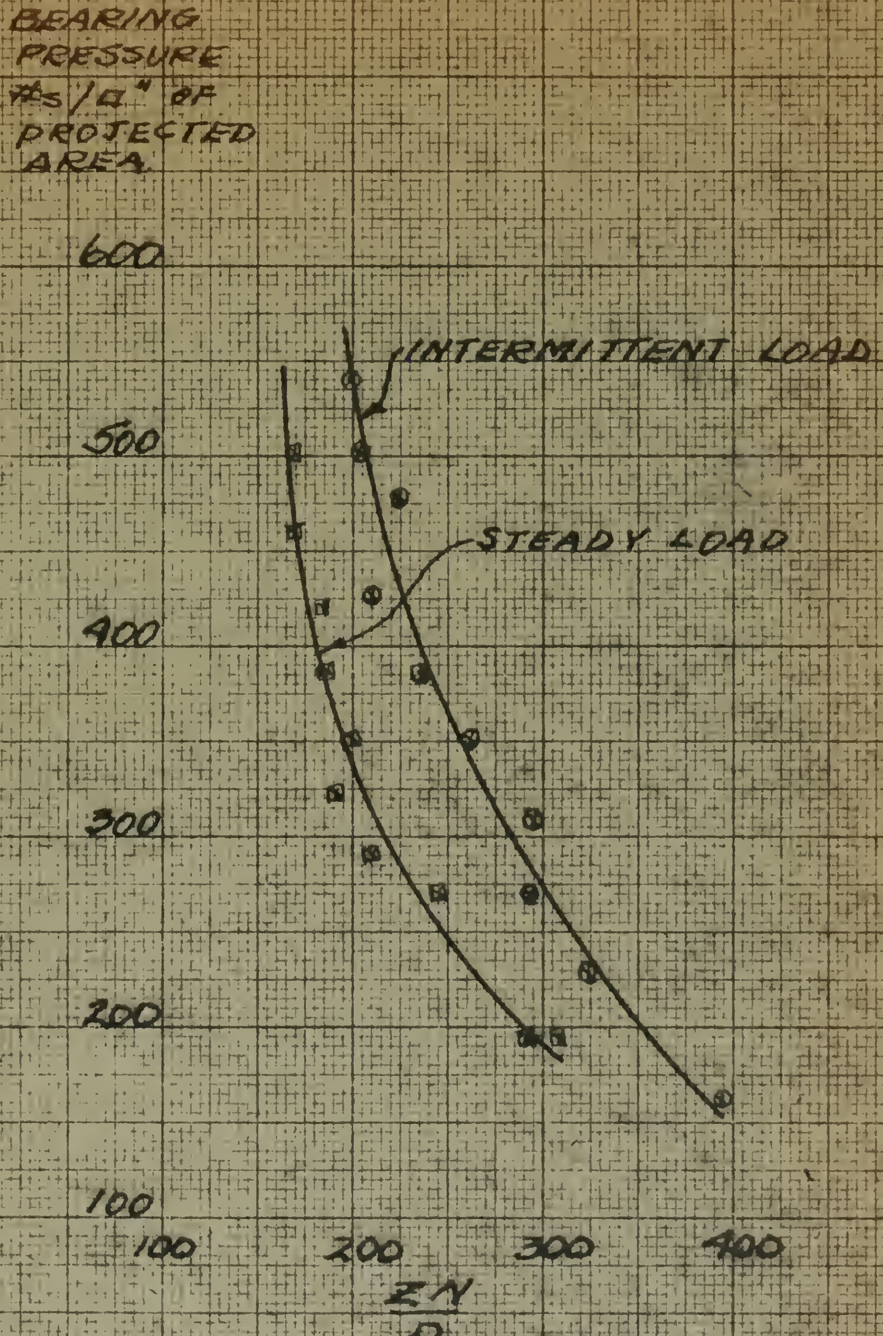
PLATE 3

ASSEMBLY VIEW OF MACHINE, OPERATING AND SPEED CONTROL EQUIPMENT



BEARING PRES 280#/□" OF PROJ AREA

FIG. 1.



Z = ABSOLUTE VISCOSITY IN CENTIPOISES
N = RPM OF JOURNAL
P = BEARING PRESSURE IN #s/□" OF PROJECTED AREA.

FIG. 2

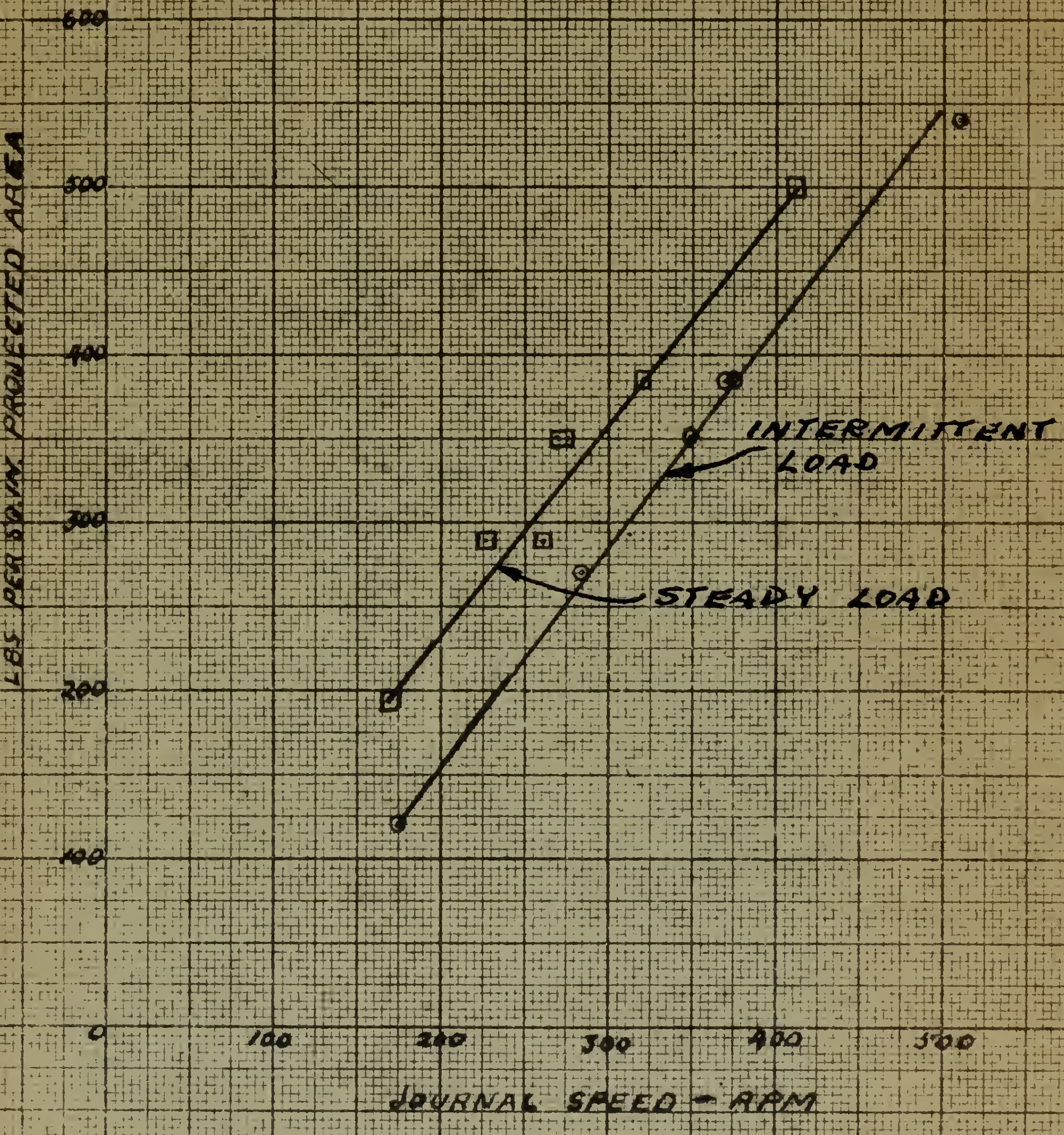

BEARING PRESSURE
LBS PER SQ IN. PROJECTED AREA
600
500
400
300
200
100
0
100
200
300
400
500
INTERMITTENT LOAD
STEADY LOAD
JOURNAL SPEED - RPM

FIG. 2 a

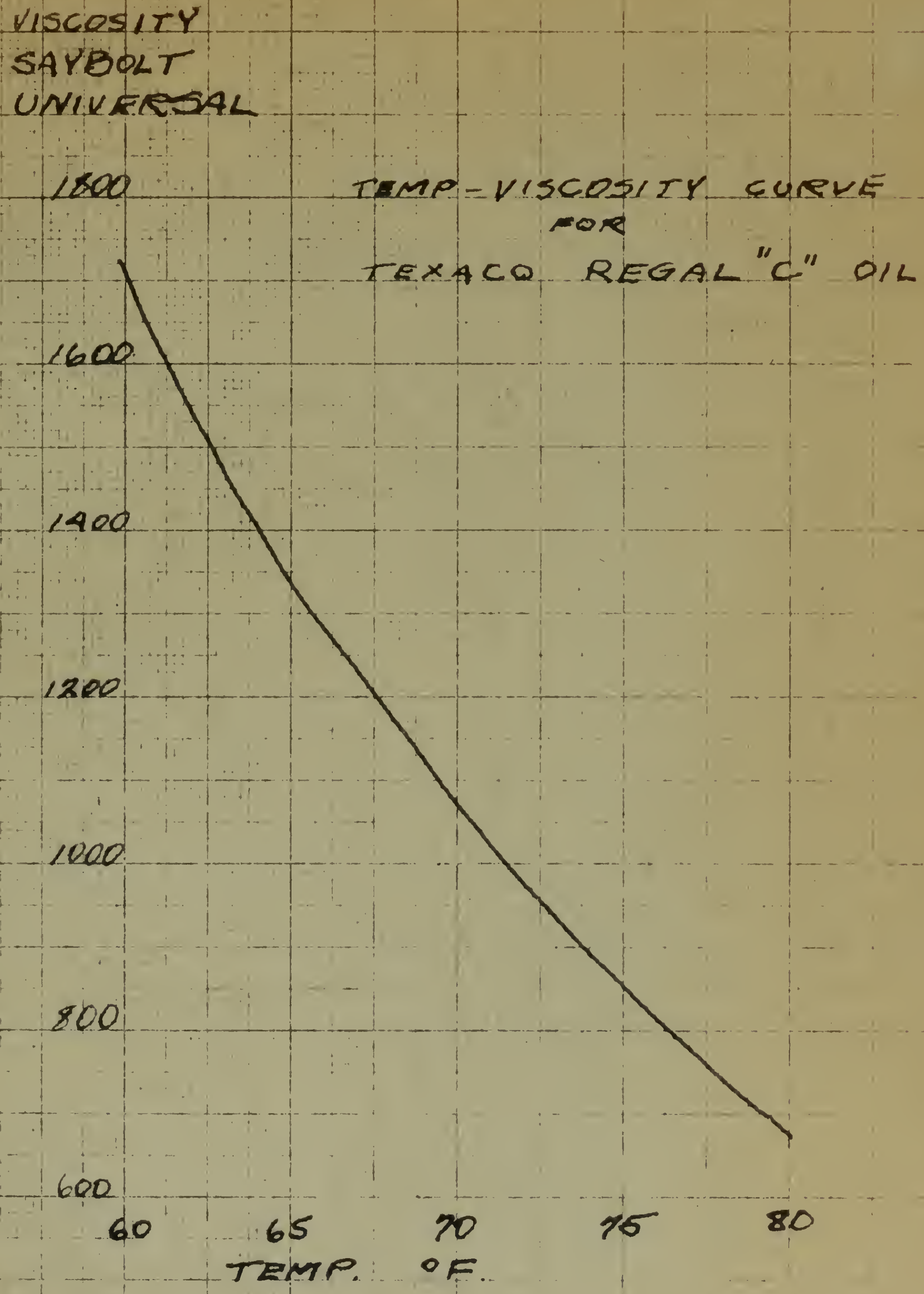
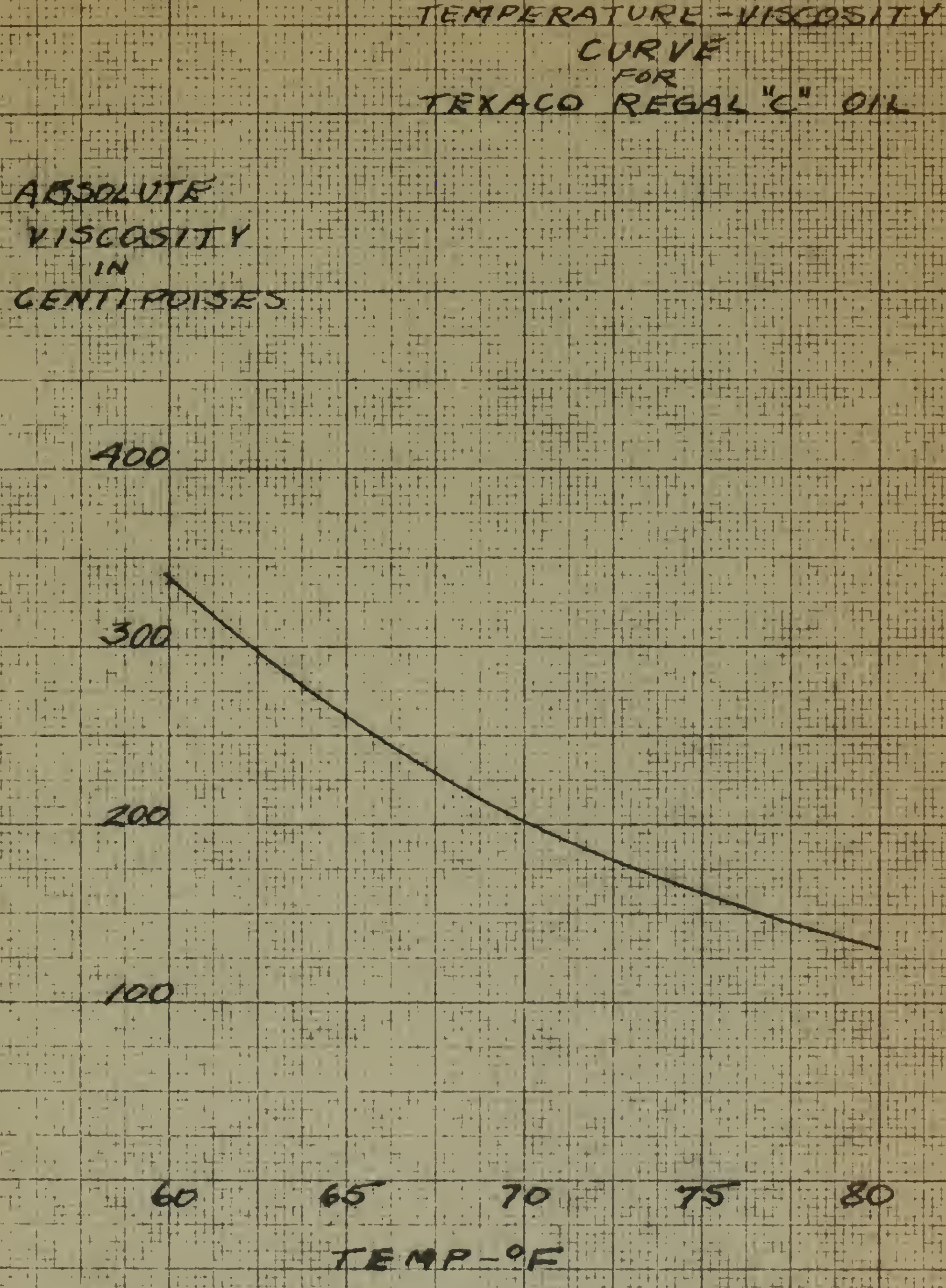

VISCOSITY
SAYBOLT
UNIVERSAL
TEMP - VISCOSITY CURVE
FOR
TEXACO REGAL "C" OIL
1800
1600
1400
1200
1000
800
600
60
65
70
75
80
TEMP. °F.

FIG. 3



TEMPERATURE-VISCOSITY CURVE FOR TEXACO REGAL "C" OIL

FIG. 3A

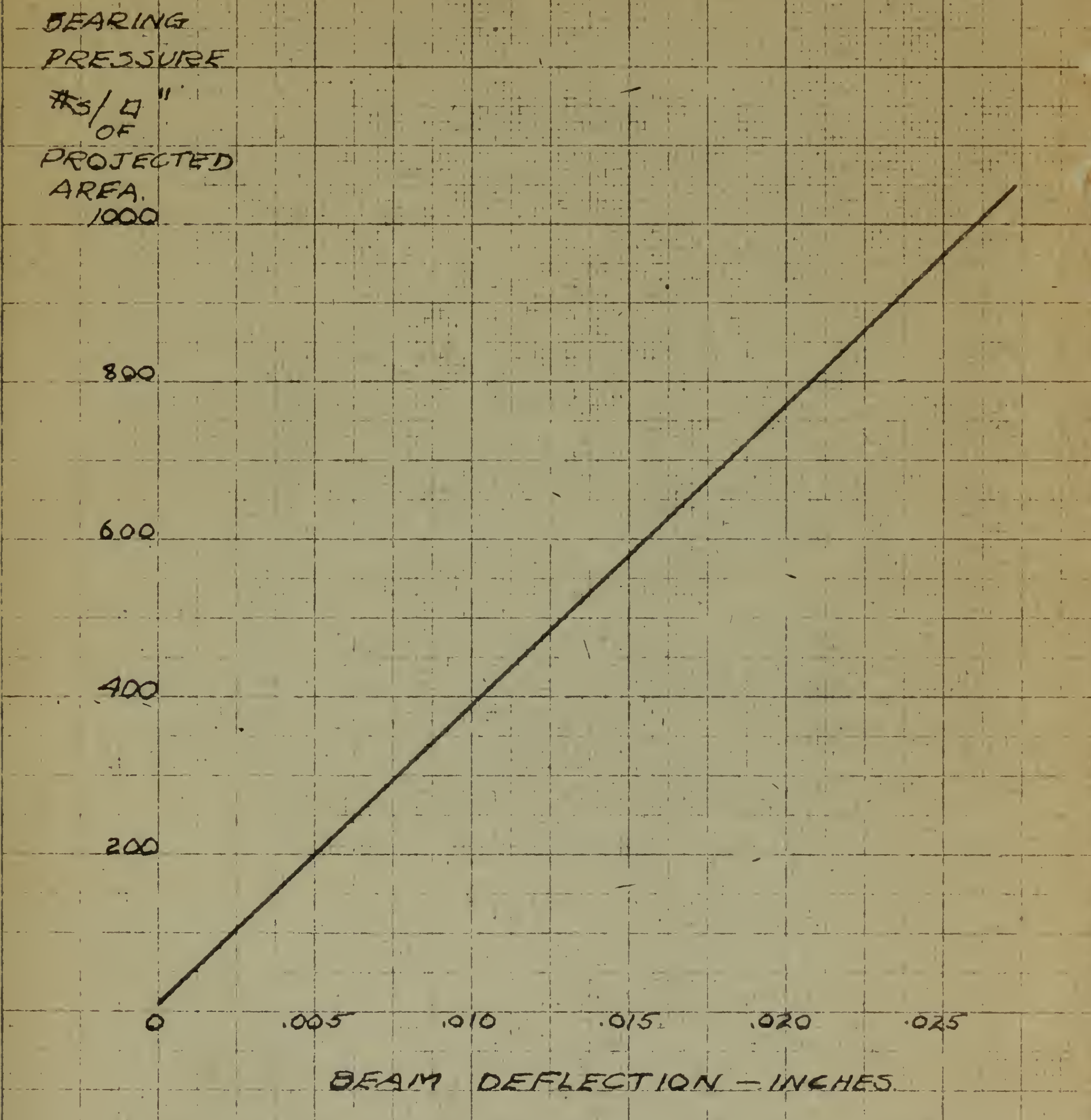
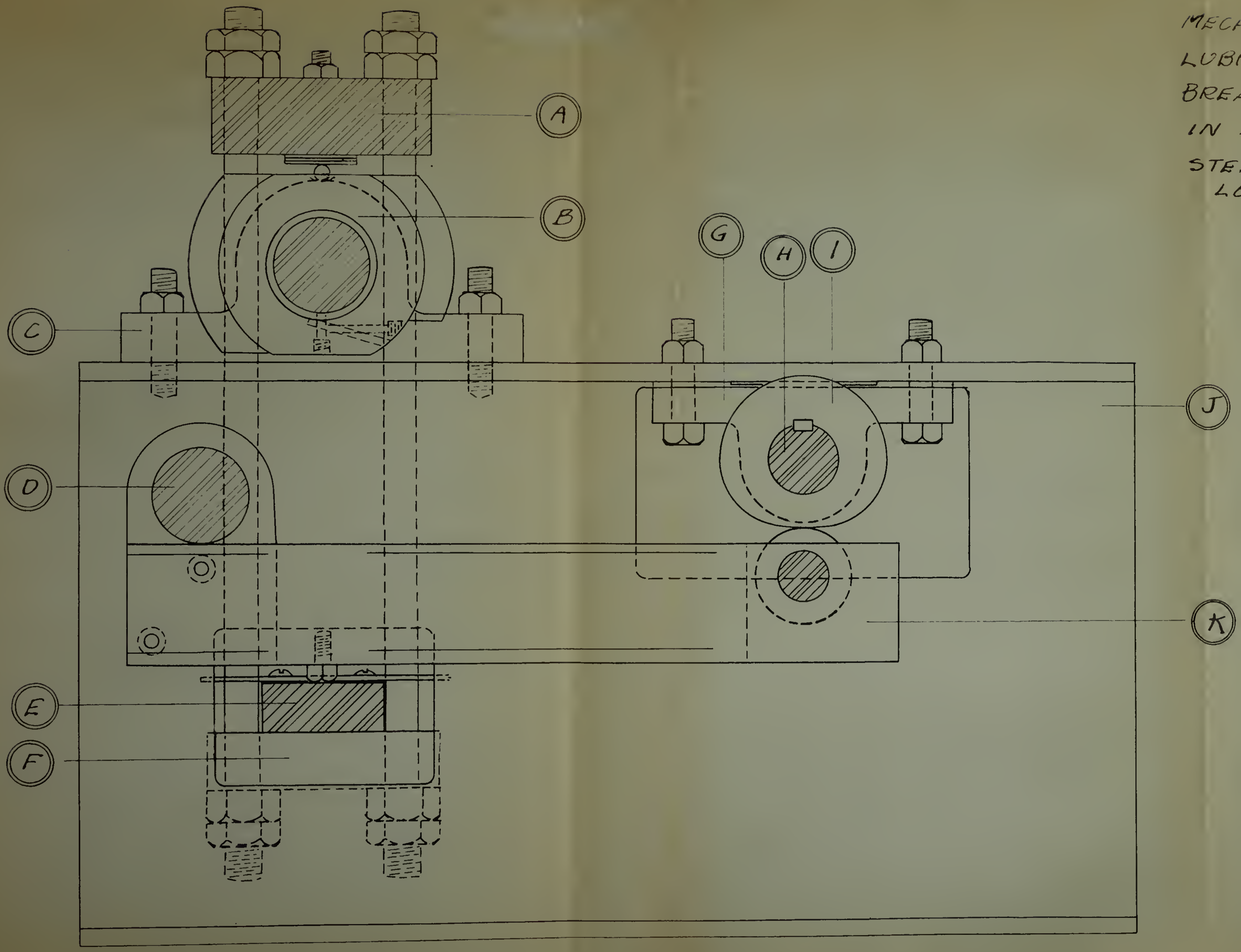

BEARING PRESSURE #s/□" OF PROJECTED AREA.
1000
800
600
400
200
0
.005
.010
.015
.020
.025
BEAM DEFLECTION – INCHES.

FIG. 4.

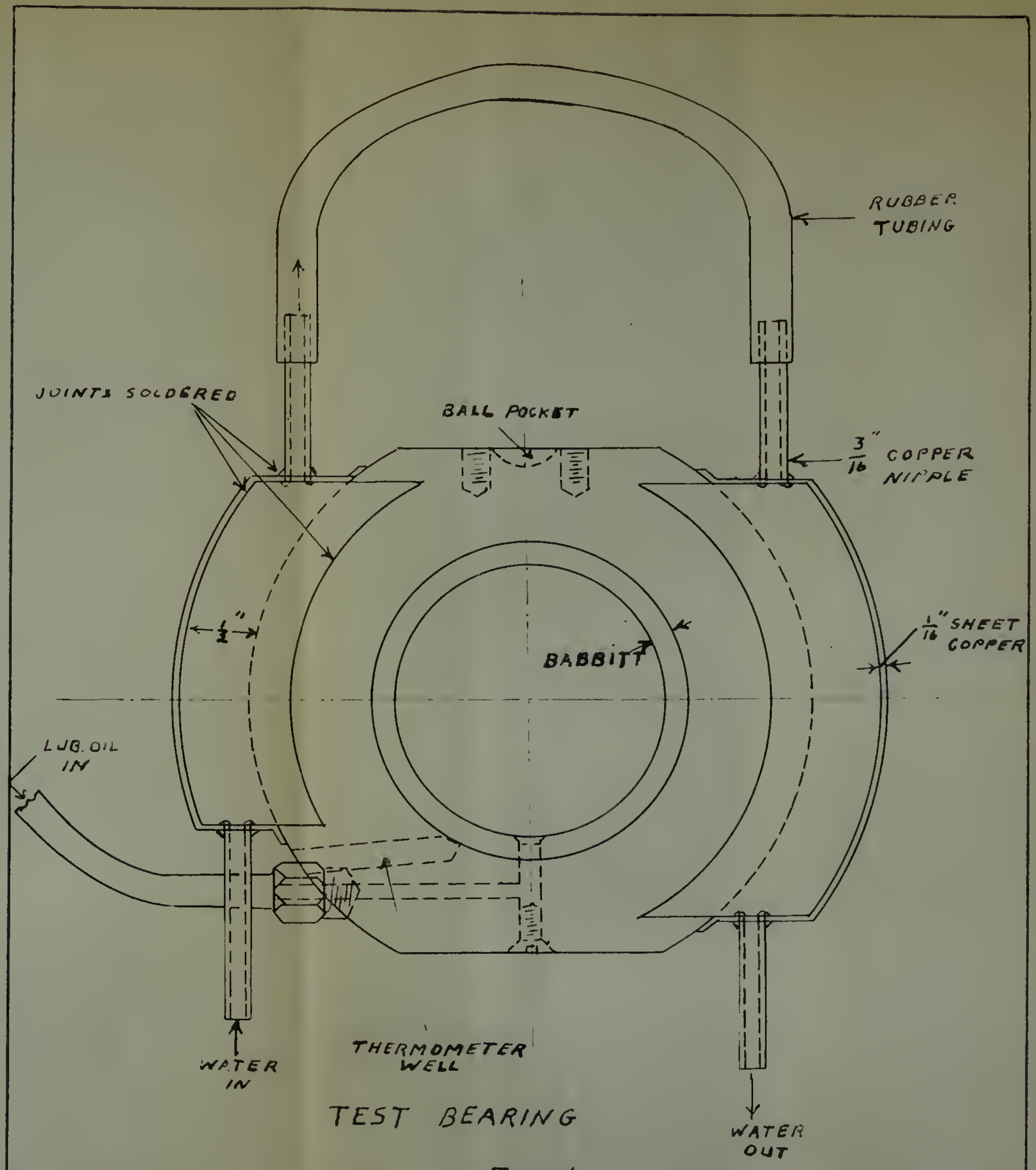
MECHANISM FOR MEASURING
LUBRICATING OIL FILM
BREAK-DOWN PRESSURES
IN A BEARING UNDER
STEADY AND INTERMITTENT
LOADING

A.D. BARNES.
D.N. CONE.



KEY TO DIAGRAM	
DES	PART
A	STRONG BACK
B	TEST BEARING
C	SUPPORT BEARINGS
D	FULCRUM SHAFT
E	LOAD BAR
F	LOAD BAR SUPPORT
G	CAMSHAFT BEARING
H	CAM SHAFT
I	LOADING CAM
J	FRAME I-BEAM
K	LEVER BAR

FIG. 5.



TEST BEARING

FIG. 6

AUG 31 BINDERY

6311

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C7 Cone

AUTHOR
An investigation of lubricating oil film ...
TITLE

DATE LOANED	BORROWER'S NAME	DATE RETURNED
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C7 Cone
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